VISIT…

SECTION 19

THREADED COMPONENTS

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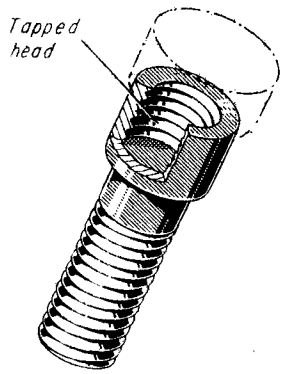
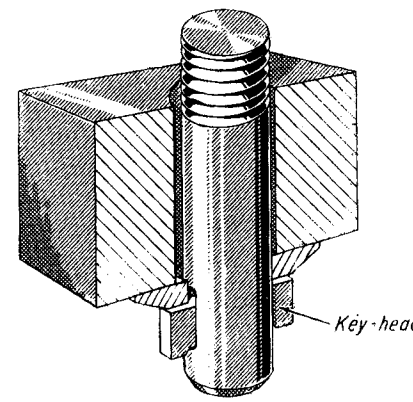
Getting the Most From Screws

Special jobs often call for special screw arrangements; here are some examples of how this busy fastener can perform.

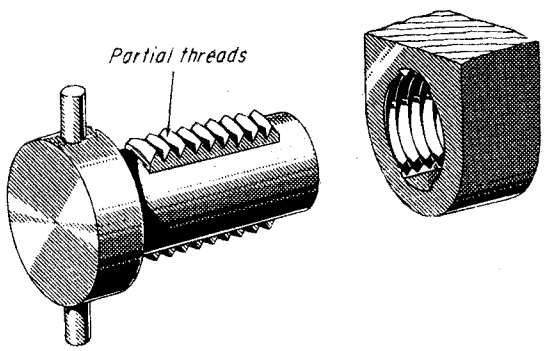
Federico Strasser



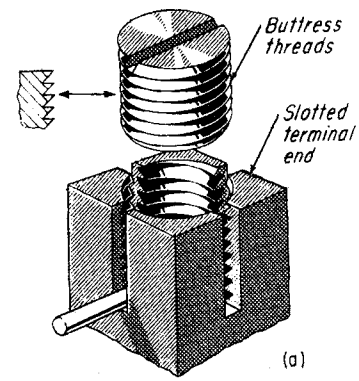
1 TAPPED HEAD LETS EXTENSIONS BE ADDED.



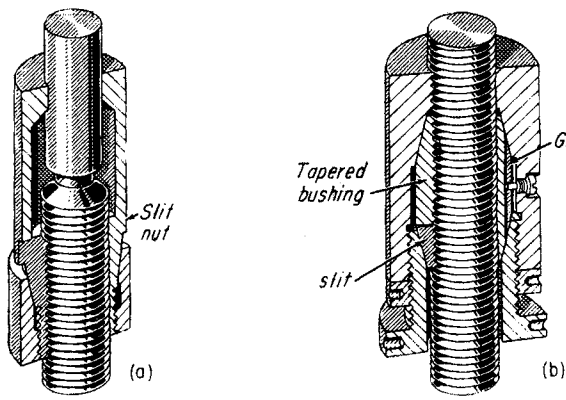
2 KEY-TYPE HEAD PROVIDES QUICK-RELEASE FEATURE.



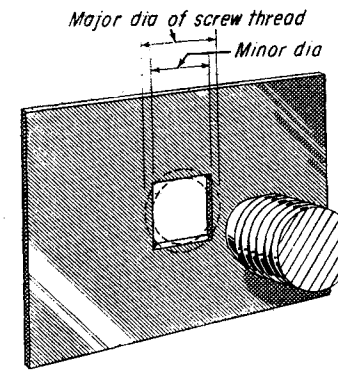
4 PARTIAL THREADS ASSEMBLE FAST, DON'T WORK LOOSE.



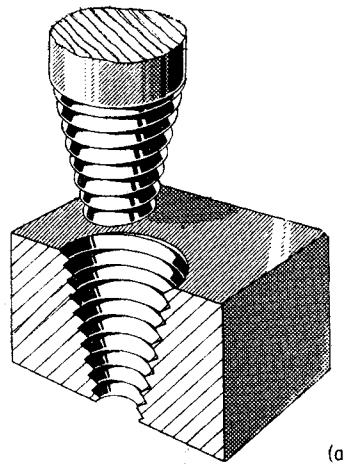
5 BUTTRESS THREADS PREVENT (a) radial forces from opening slotted ends; otherwise (b) a reinforcing sleeve is needed.



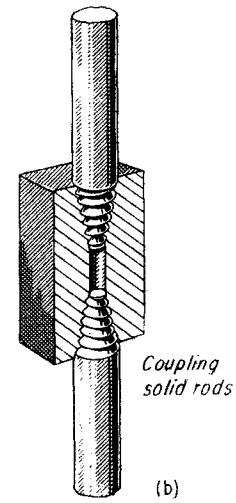
7 SLIT NUT (a) and tapered bushing (also slit) (b) allows backlash-free adjustment.



8 SQUARE HOLE for light metal or plastic substitutes well for threaded holes.

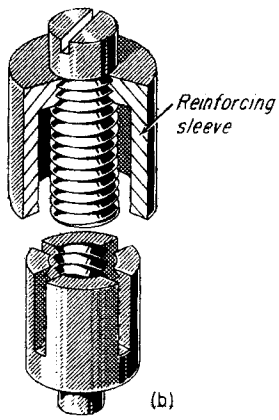


(a)

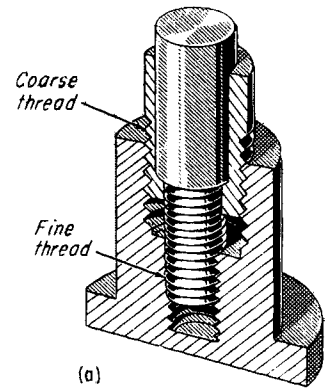


(b)

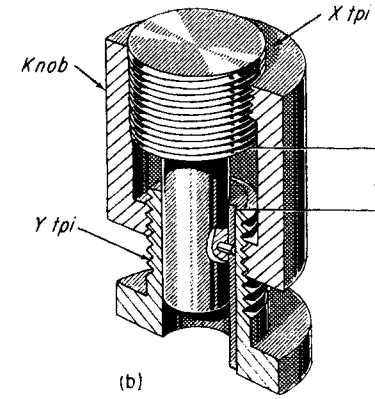
3 **TAPERED SCREWS ASSEMBLE AND RELEASE FAST, BUT WORK LOOSE EASILY.**



(b)

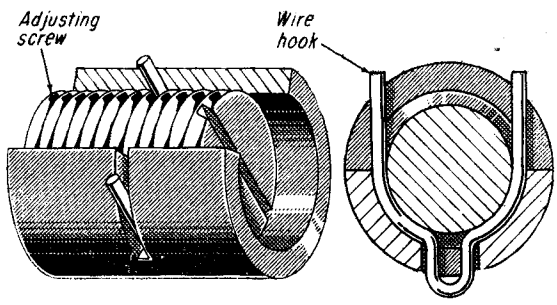


(a)

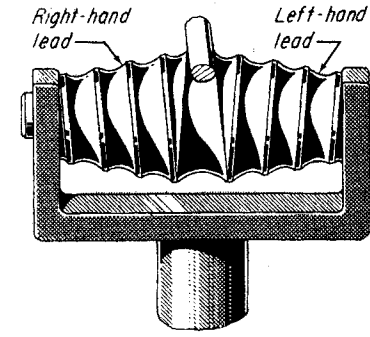


(b)

6 **DIFFERENTIAL THREADS PROVIDE (a) extra tight fastening or (b) extra small relative movement, δ, per revolution of knob.**



9 **WIRE HOOK provides single-thread grip for low-cost device.**

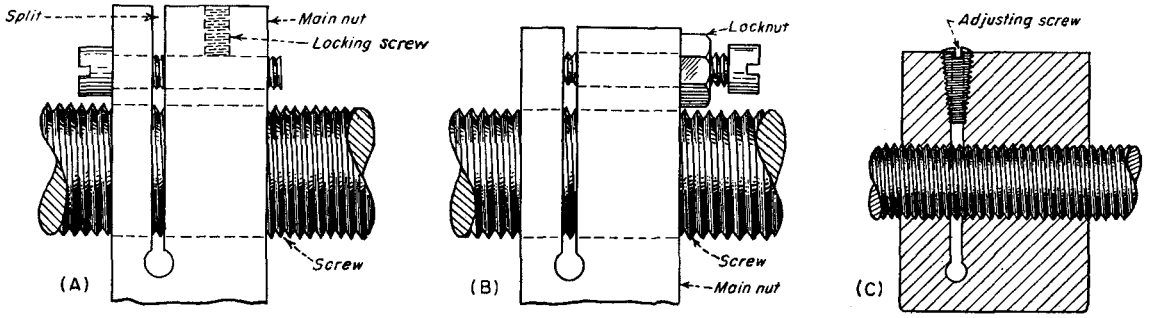


10 **DOUBLE SCREW for wire guide or follower always leads wire to center.**

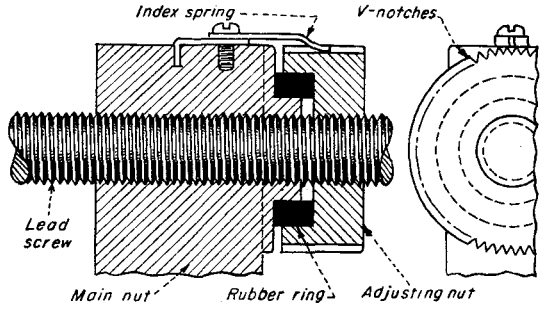
How to Provide for Backlash in Threaded Parts

These illustrations are based on two general methods of providing for lost motion or backlash. One allows for relative movement of the nut and screw in the plane parallel to the thread axis; the other method involves a radial adjustment to compensate for clearance between sloping faces of the threads on each element.

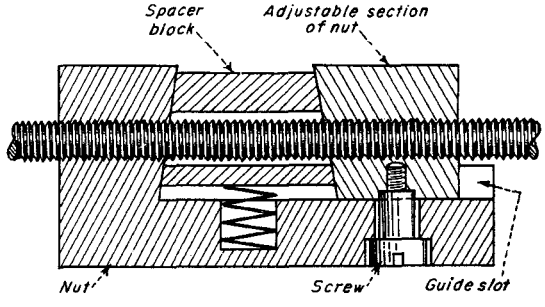
Clifford T. Bower



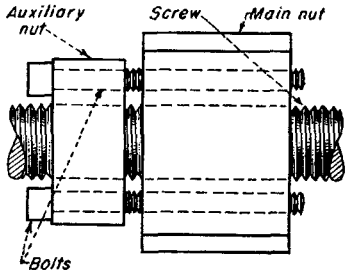
THREE METHODS of using slotted nuts. In *(A)*, nut sections are brought closer together to force left-hand nut flanks to bear on right-hand flanks of screw thread and vice versa. In *(B)*, and *(C)* nut sections are forced apart for same purpose.



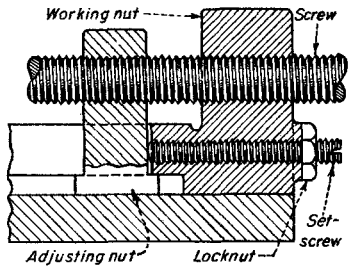
AROUND THE PERIPHERY of the backlash-adjusting nut are "v" notches of small pitch which engage the index spring. To eliminate play in the lead screw, adjusting nut is turned clockwise. Spring and adjusting nut can be calibrated for precise use.



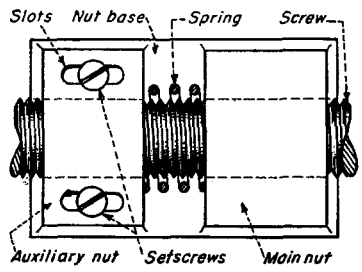
SELF-COMPENSATING MEANS of removing backlash. Slot is milled in nut for an adjustable section which is locked by a screw. Spring presses the tapered spacer block upwards, forcing the nut elements apart, thereby taking up backlash.



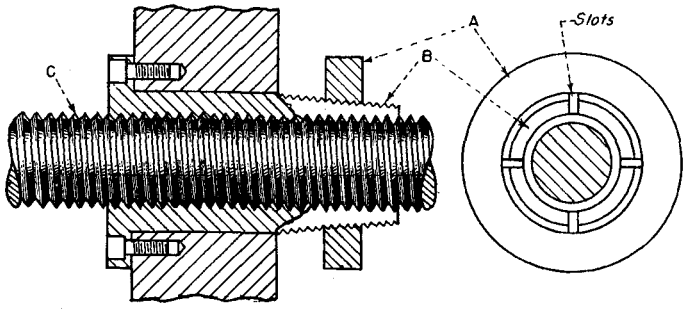
MAIN NUT is integral with base attached to part moved by screw. Auxiliary nut is positioned one or two pitches from main nut. The two are brought closer together by bolts which pass freely through the auxiliary nut.



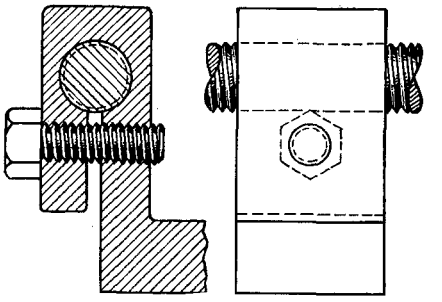
ANOTHER WAY to use an auxiliary or adjusting nut for axial adjustment of backlash. Relative movement between the working and adjusting nuts is obtained manually by the set screw which can be locked in place as shown.



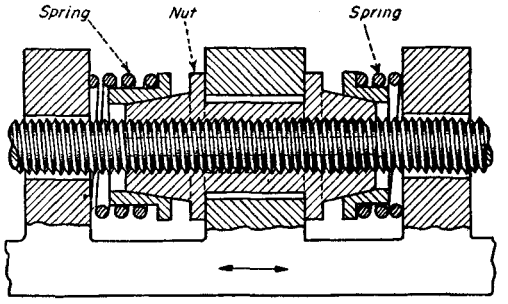
COMPRESSION SPRING placed between main and auxiliary nuts exerts force tending to separate them and thus take up slack. Set screws engage nut base and prevent rotation of auxiliary nut after adjustment is made.



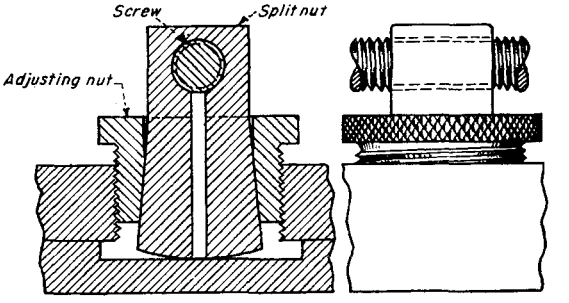
NUT *A* IS SCREWED along the tapered round nut, *B*, to eliminate backlash or wear between *B* and *C*, the main screw, by means of the four slots shown.



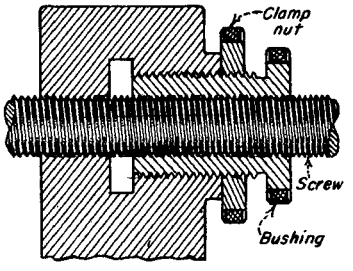
ANOTHER METHOD of clamping a nut around a screw to reduce radial clearance.



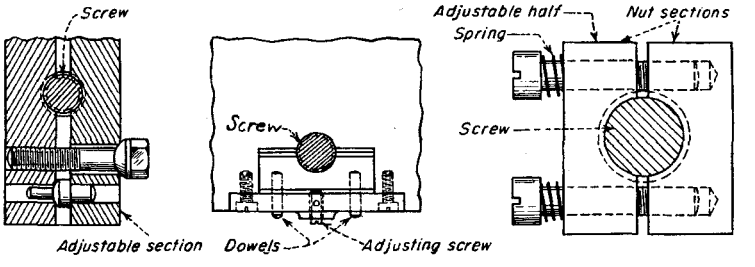
AUTOMATIC ADJUSTMENT for backlash. Nut is flanged on each end, has a square outer section between flanges and slots cut in the tapered sections. Spring forces have components which push slotted sections radially inward.



SPLIT NUT is tapered and has a rounded bottom to maintain as near as possible a fixed distance between its seat and the center line of the screw. When the adjusting nut is tightened, the split nut springs inward slightly.



CLAMP NUT holds adjusting bushing rigidly. Bushing must have different pitch on outside thread than on inside thread. If outer thread is the coarser one, a relatively small amount of rotation will take up backlash.

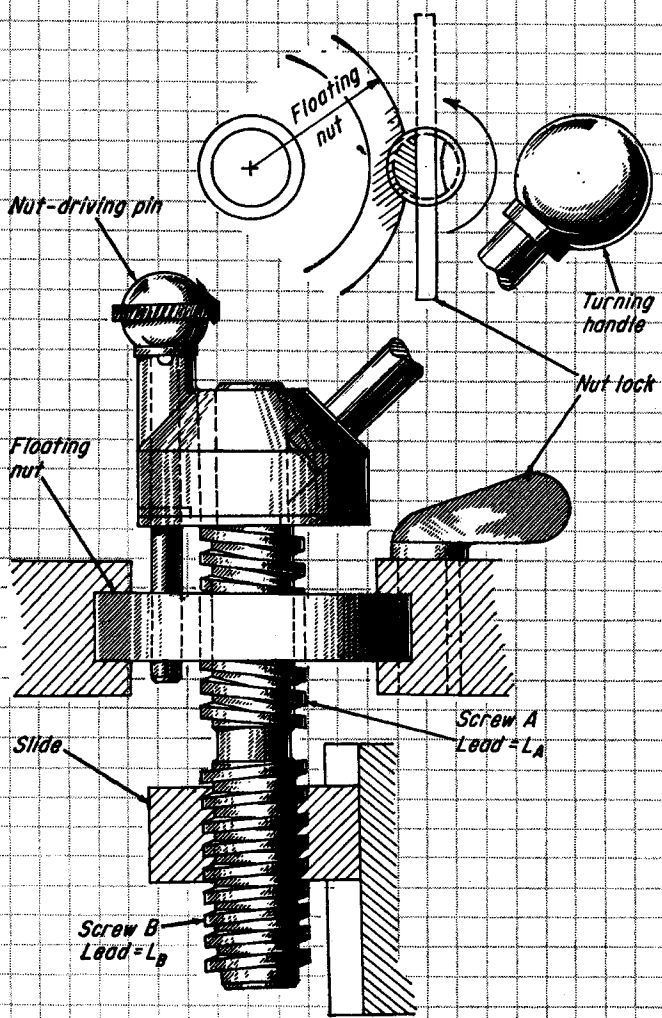


TYPICAL CONSTRUCTIONS based on the half nut principle. In each case, the nut bearing width is equal to the width of the adjustable or inserted slide piece. In the sketch at the extreme left, the cap screw with the spherical seat provides for adjustments. In the center sketch, the adjusting screw bears on the movable nut section. Two dowels insure proper alignment. The third illustration is similar to the first except that two adjusting screws are used instead of only one.

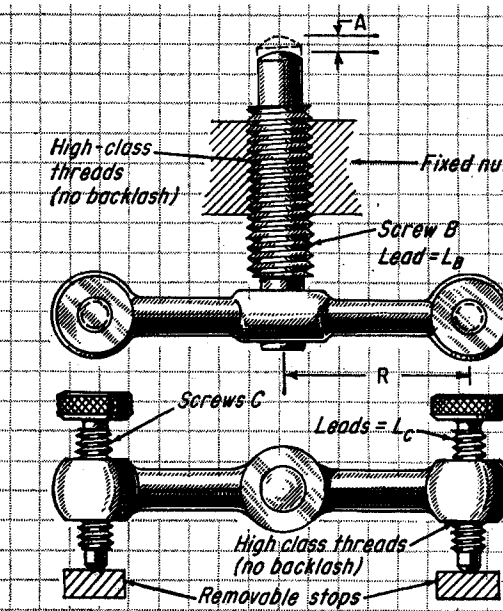
7 Special Screw Arrangements

How differential, duplex, and other types of screws can provide slow and fast feeds, minute adjustments, and strong clamping action.

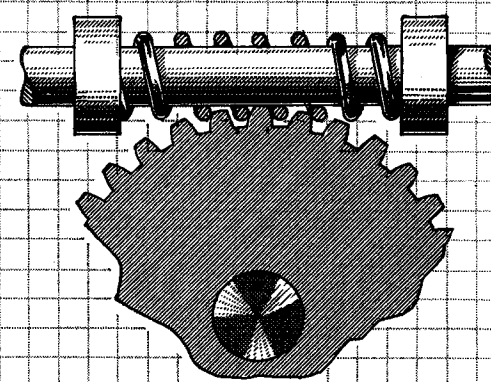
Louis Dodge



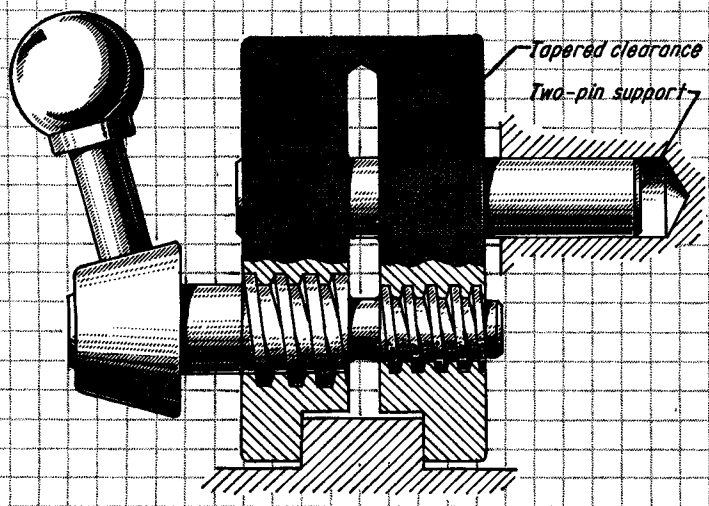
1 **RAPID AND SLOW FEED.** With left- and right-hand threads, slide motion with nut locked equals L_A plus L_B per turn; with nut floating, slide motion per turn equals L_B. Get extremely fine feed with rapid return motion when threads are differential.



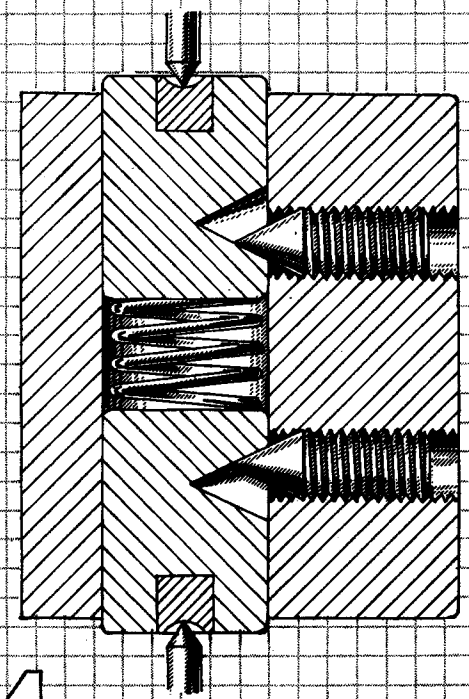
2 **EXTREMELY SMALL MOVEMENTS.** Microscopic measurements, for example, are characteristic of this arrangement. Movement A is equal to $N(L_B \times L_C)/2\pi R$, where N equals number of turns of screw C.



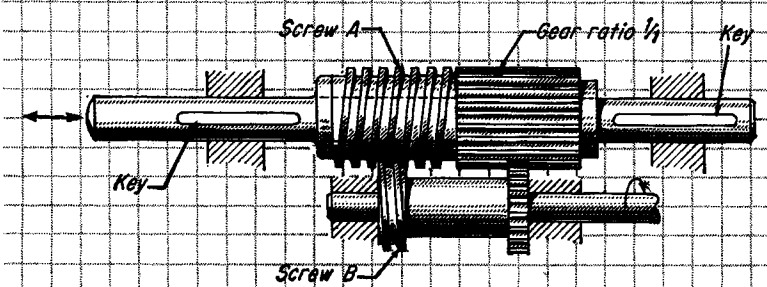
5 **SHOCK ABSORBENT SCREW.** When springs coiled as shown are used as worm drives for light loads, they have the advantage of being able to absorb heavy shocks.



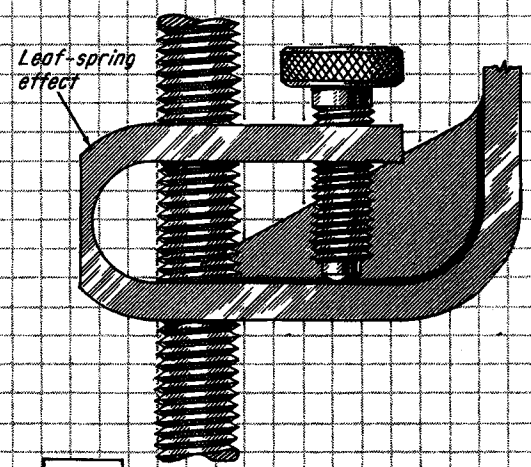
3 **DIFFERENTIAL CLAMP.** This method of using a differential screw to tighten clamp jaws combines rugged threads with high clamping power. Clamping pressure, $P = Te/[R(\tan \varphi + \tan \alpha]$, where T = torque at handle, R = mean radius of screw threads, φ = angle of friction (approx. 0.1), α = mean pitch angle of screw, and e = efficiency of screw (generally about 0.8).



4 **BEARING ADJUSTMENT.** This screw arrangement is a handy way of providing for bearing adjustment and overload protection.



6 **HIGH REDUCTION** of rotary motion to fine linear motion is possible here. Arrangement is for low forces. Screws are left and right hand. $L_A = L_B$ plus or minus a small increment. When $L_B = 1/10$ and $L_A = 1/10.05$ the linear motion of screw A will be 0.05 in. per turn. When screws are the same hand, linear motion equals $L_A + L_B$.



7 **BACKLASH ELIMINATION.** The large screw is locked and all backlash is eliminated when the knurled screw is tightened — finger torque is sufficient.

20 Dynamic Applications for Screw Threads

Have you forgotten how simply, and economically, screw threads can be made into dynamic members of a linkage? Here are some memory-joggers, plus suggestions for simplified nuts, threads and nut guides.

Kurt Rabe

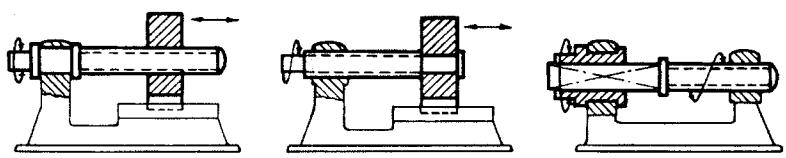
You need a threaded shaft, a nut . . . plus some way for one of these members to rotate without translating and the other to translate without rotating. That's all. Yet these simple components can do practically all of the adjusting, setting, or locking used in design.

Most such applications have low-precision requirements. That's why the thread may be a coiled wire or a twisted strip; the nut may be a notched ear on a shaft or a slotted disk. Standard screws and nuts right off your supply shelves can often serve at very low cost.

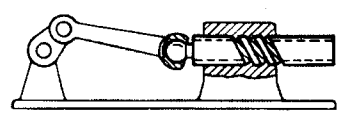
Here are the basic motion transformations possible with screw threads (Fig 1):

- transform rotation into linear motion or reverse (A),
- transform helical motion into linear motion or reverse (B),
- transform rotation into helical motion or reverse (C).

Of course the screw thread may be combined with other components: in a 4-bar linkage (Fig 2), or with multiple screw elements for force or motion amplification.



1 MOTION TRANSFORMATIONS of a screw thread include: rotation to translation (A), helical to translation (B), rotation to helical (C). Any of these is reversible if the thread is not self-locking (see screw-thread mathematics on following page—thread is reversible when efficiency is over 50%).



2 STANDARD 4-BAR LINKAGE has screw thread substituted for slider. Output is helical rather than linear.

A Review of Screw-Tread Mathematics

a — friction angle, $\tan a = f$
r — mean radius of thread
= ½ (root radius + outside radius), in inches
l — lead, thread advance in one revolution, in.
b — lead angle, $\tan b = 1/2\pi r$, deg
f — friction coefficient
P — equivalent driving force at radius r from screw axis, lb
L — axial load, lb
e — efficiency
c — half angle between thread faces, deg

SQUARE THREADS:

$$P = L \tan (b \pm a) = L \frac{(l \pm 2\pi rf)}{(2\pi r \mp fl)}$$

Where upper signs are for motion opposed in direction to L. Screw is self-locking when $b \leqq a$.

$$e = \frac{\tan b}{\tan (b + a)} \quad \text{(motion opposed to } L\text{)}$$

$$e = \frac{\tan (b - a)}{\tan b} \quad \text{(motion assisted by } L\text{)}$$

V THREADS:

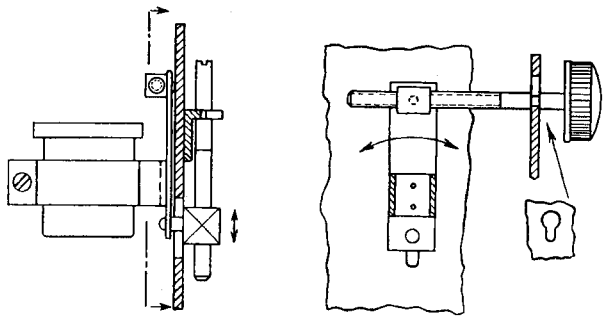
$$P = L \frac{(l \pm 2\pi rf \sec c)}{(2\pi r \mp lf \sec c)}$$

$$e = \frac{\tan b\,(1 - f \tan b \sec c)}{(\tan b + f \sec c)} \quad \text{(motion opposed to } L\text{)}$$

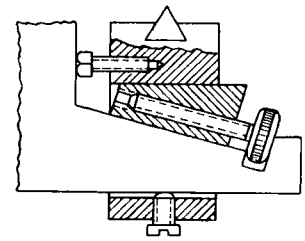
$$e = \frac{\tan b - f \sec c}{\tan b\,(1 + f \tan b \sec c)} \quad \text{(motion assisted by } L\text{)}$$

For more detailed analysis of screw-thread friction forces, see Marks *Mechanical Engineers' Handbook*, McGraw-Hill Book Co.

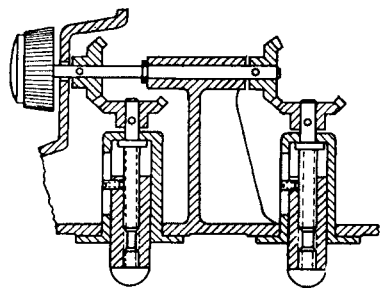
Rotation to Translation



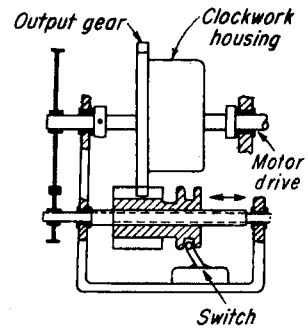
3 TWO-DIRECTIONAL LAMP ADJUSTMENT with screwdriver to move lamp up and down. Knob adjust (right) rotates lamp about pivot.



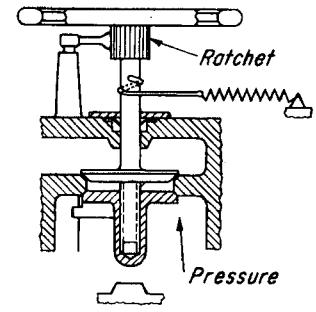
4 KNIFE-EDGE BEARING is raised or lowered by screw-driven wedge. Two additional screws locate the knife edge laterally and lock it.



5 SIDE-BY-SIDE ARRANGEMENT of tandem screw threads gives parallel rise in this height adjustment for projector.

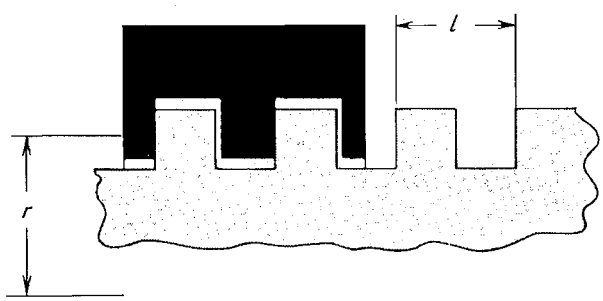


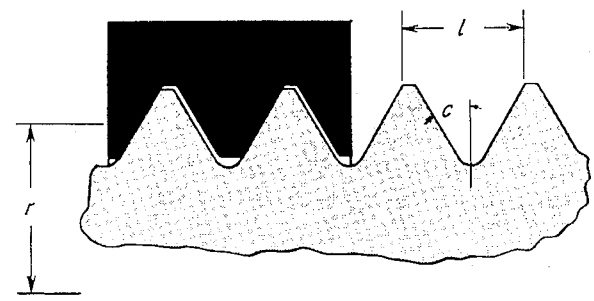
6 AUTOMATIC CLOCKWORK is kept wound tight by electric motor turned on and off by screw thread and nut. Note motor drive must be self-locking or it will permit clock to unwind as soon as switch turns off.



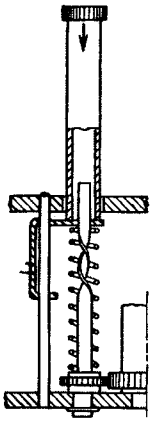
7 VALVE STEM has two oppositely moving valve cones. When opening, the upper cone moves up first, until it contacts its stop. Further turning of the valve wheel forces the lower cone out of its seat. The spring is wound up at the same time. When the ratchet is released, spring pulls both cones into their seats.

A Review of Screw-Tread Mathematics Continued

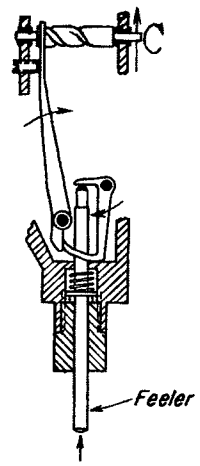




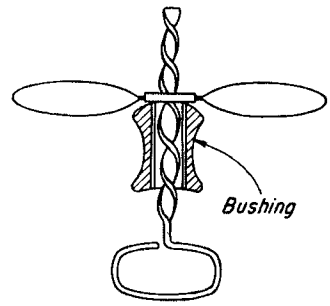
Translation to Rotation



8 A METAL STRIP or square rod may be twisted to make a long-lead thread, ideal for transforming linear into rotary motion. Here a push-button mechanism winds a camera. Note that the number of turns or dwell of output gear is easily altered by changing (or even reversing) twist of the strip.

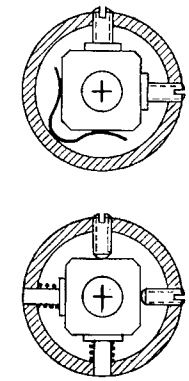


9 FEELER GAGE has its motion amplified through a double linkage and then transformed to rotation for dial indication.

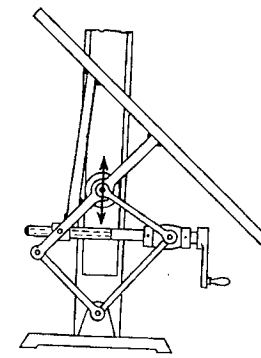


10 THE FAMILIAR flying propeller-toy is operated by pushing the bushing straight up and off the thread.

Self-Locking

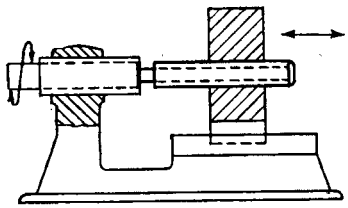


11 HAIRLINE ADJUSTMENT for a telescope, with two alternative methods of drive and spring return.

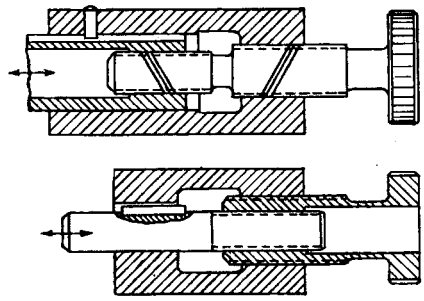


12 SCREW AND NUT provide self-locking drive for a complex linkage.

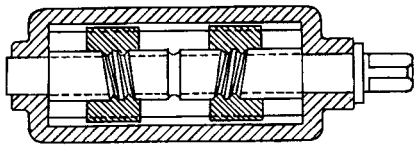
Double Threading



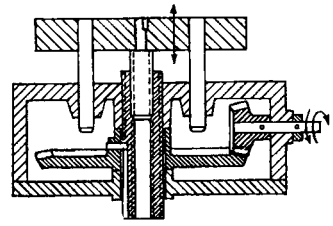
13 DOUBLE-THREADED SCREWS, when used as differentials, provide very fine adjustment for precision equipment at relatively low cost.



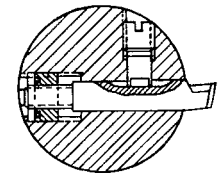
14 DIFFERENTIAL SCREWS can be made in dozens of forms. Here are two methods: above, two opposite-hand threads on a single shaft; below, same hand threads on independent shafts.



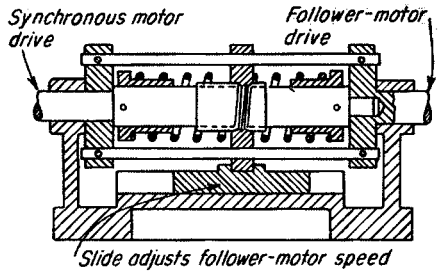
15 OPPOSITE-HAND THREADS make a high-speed centering clamp out of two moving nuts.



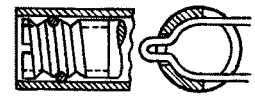
16 MEASURING TABLE rises very slowly for many turns of the input bevel gear. If the two threads are 1½—12 and ¾—16, in the fine-thread series, table will rise approximately 0.004 in. per input-gear revolution.



17 LATHE TURNING TOOL in drill rod is adjusted by differential screw. A special double-pin wrench turns the intermediate nut, advancing the nut and retracting the threaded tool simultaneously. Tool is then clamped by setscrew.

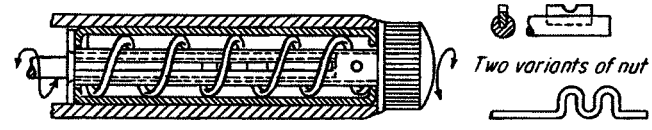


18 ANY VARIABLE-SPEED MOTOR can be made to follow a small synchronous motor by connecting them to the two shafts of this differential screw. Differences in number of revolutions between the two motors appear as motion of the traveling nut and slide so an electrical speed compensation is made.



19 (left) **A WIRE FORK** is the nut in this simple tube-and screw design.

20 (below) **A MECHANICAL PENCIL** includes a spring as the screw thread and a notched ea or a bent wire as the nut.



EDITOR'S NOTE: For other solutions to adjusting, setting, and locking problems in translating motion, see:

10 Ways to Employ Screw Mechanisms, May 26 '58, p 80. Shows applications in terms of three basic components—actuating member, threaded device, and sliding device.

5 Cardan-gear Mechanisms. Sep 28 '59, p 66. Gearing arrangements that convert rotation into straight-line motion.

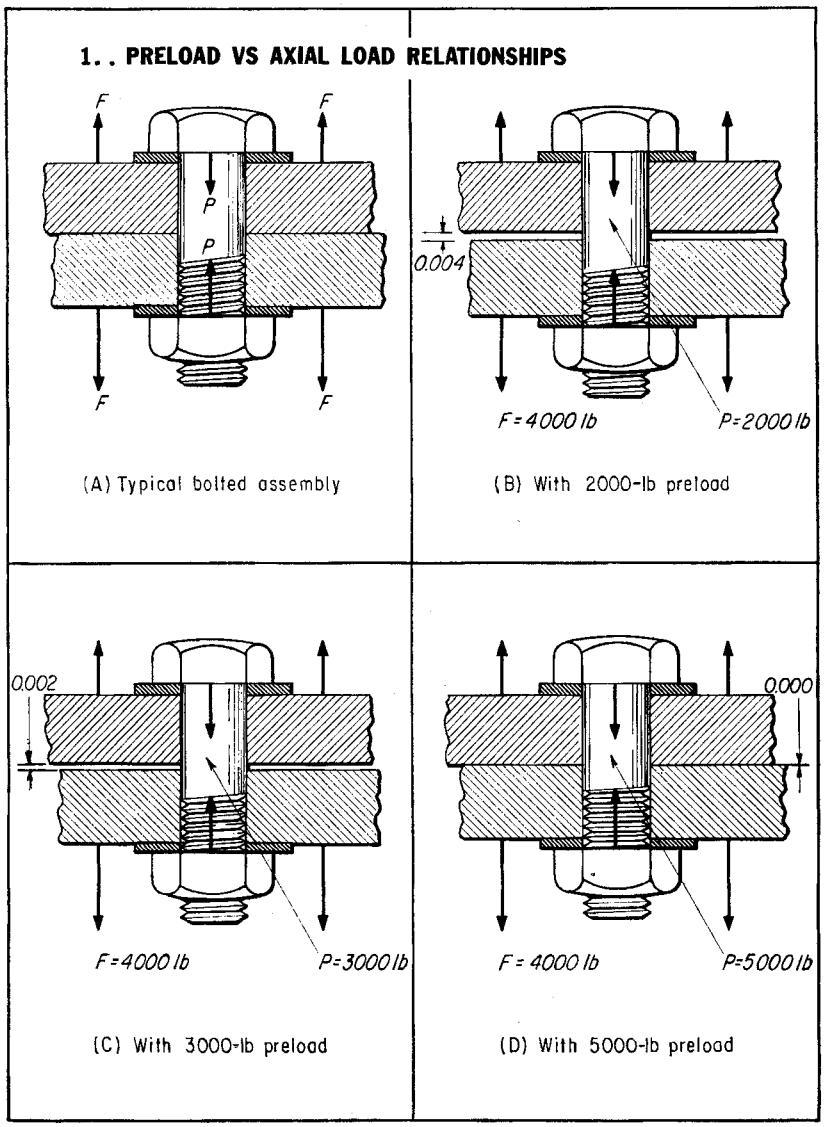
5 Linkages for Straight-line Motion, Oct 12 '59, p 86. Linkages that convert rotation into straight-line motion.

For REPRINT of above article, just check P35 on one of the Reader Service cards bound in this issue.

Preloading of Bolts

For optimum tension-bolt design, the bolts must be preloaded to a specific amount. Here are charts which quickly give the data you need.

Bernie J. Cobb



THE rule is, always tighten a bolted assembly to the maximum permissible preload. The benefits are twofold: 1) You extend the fatigue life, by reducing or eliminating fluctuations in stress. 2) You can design a smaller, lighter, less costly assembly for a given tensile load.

Although other methods have been published for computing the necessary data, the design charts offered here are both accurate and simple to use. They show a direct relationship for the four key factors: bolt diameter, bolt strength, tightening torque, and preload. When any two of these factors are given, the charts produce the other two.

The stress equation on which the charts are based is derived from the currently accepted maximum strain theory. The data in these charts have simplified critical problems in missile design and have been successfully applied to the design of solid propellant rocket motors, where the bolts must absorb the blow-off load exerted by internal pressure of the motor against the nozzle and igniter, maintain a positive pressure on the gasket between the flanges, and be as lightweight as possible.

How preload works

A bolted unit is shown in Fig 1. The bolt is tightened (in A) to produce a preload, *P*. The applied axial load, *F*, may be the result of internal pressure (the plates may then represent flanges), or of external loads (one of the plates may be a bracket, the other a machine frame).

If a tightening torque is applied to the nut to provide a 2000-lb preload,

as in B, and if the applied load varies from zero to a maximum of 4000 lb, the bolt will experience a cyclic load change of 2000 lb which will contribute to fatigue failure. There will also be a separation between plates of, say, 0.004 in. (This is a function of bolt diameter and material.)

If the preload is increased to 3000 lb, as in C, the load variations will be reduced to 1000 lb, which will increase fatigue life and reduce plate separation under load by 50%, to 0.002 in. If the preload is now increased to 5000 lb, as in D, there will be no separation of the plates under load, and the load variations will be reduced to zero. Fatigue failure should not occur.

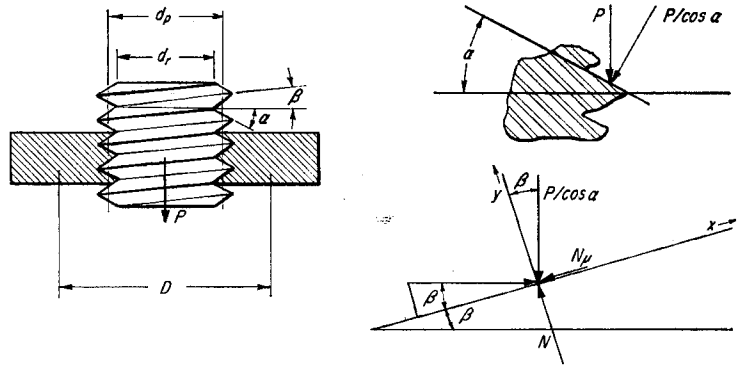
The important information to know in the above analysis is the maximum preload that can be safely applied to a particular bolt, and how much tightening torque must be applied to obtain the maximum preload.

Maximum preload

A value accepted by many designers and recommended by Kent (*Mechanical Engineer's Handbook*, 12th ed), is that the prestress must not exceed 80% of the bolt-proof strength. The resulting 20% margin of safety allows for variations in applied torque and for the friction forces that may occur. Proof strength is defined as 85% of

SYMBOLS

d_p = Pitch diameter of screw threads, in.
d_r = Minor diameter of screw threads, in.
D = Mean bearing diameter of nut, in.
$m = \dfrac{\tan(\beta+\phi)}{\cos\alpha}$
N = Reaction force normal to helix, lb
F = Applied axial load in bolt, lb
P = Maximum permissible preload, lb
S = Combined bolt stress, psi
S_t = Tensile stress, psi
S_s = Combined stress, psi
T = Applied tightening torque to bolt head, in.-lb
α = One half of thread profile angle, deg
β = Helix angle of thread, deg
ϵ = Poisson's ratio (0.3 for steel)
μ = Coefficient of friction between threads
ν = Coefficient of friction between nut and washer on bearing surface
ϕ = Friction angle, deg, $\tan\phi = \mu$



DERIVATION OF DESIGN EQUATIONS

Load-torque ratio

The summation of forces, left, acting on a thread along the x and y axis are

$$F\cos\beta - \sin\beta\,(P/\cos\alpha) - N\mu = 0$$

$$N - \cos\beta\,(P/\cos\alpha) - F\sin\beta = 0$$

Combining these equations and substituting $\tan\phi$ for μ results in

$$\frac{F}{P} = \frac{\tan(\phi+\beta)}{\cos\alpha} = m$$

The torque acting on the bolt is

$$T = \frac{1}{2}Fd_p + \frac{1}{2}PD\nu,\quad \frac{P}{T} = \frac{2}{D\nu + m\,d_p}$$

Stress-torque ratio

For pure tension and torsion in the bolt

$$S_t = \frac{P}{A} = \frac{4P}{\pi d_r^2},\quad S_s = \frac{16T}{\pi d_r^3}$$

Since the torque on the bolt shank $T = Fd_p/2$, and $F = Pm$, then

$$S_s = \frac{8Pm\,d_p}{\pi d_r^3},\quad \frac{S_s}{S_t} = 2m\frac{d_p}{d_r}$$

To combine these stresses, the maximum strain theory by Saint-Venant is employed which states that elastic failure occurs when the deformation normal to any plane in the body reaches a value equal to the strain at the yield stress in pure tension:

$$S = \frac{1}{2}\left[(1-\epsilon)\,S_t + (1+\epsilon)(S_t^2 + 4\,S_s^2)^{1/2}\right] \qquad (1)$$

To obtain a solution for S, the values for S_t and S_s must first be calculated. If S is greater than the yield strength of the material, then S_t or S_s must be reduced, or failure may likely occur. The yield strength in shear is usually about 60% of the tensile yield, or $S_s = 0.6S$. To modify Eq 1 so that the above relationship holds true when there is pure torsion (when $S_t = 0$), a correction factor, C, must be employed. Hence, from Eq 1 with $\epsilon = 0.3$ and $S_t = 0$:

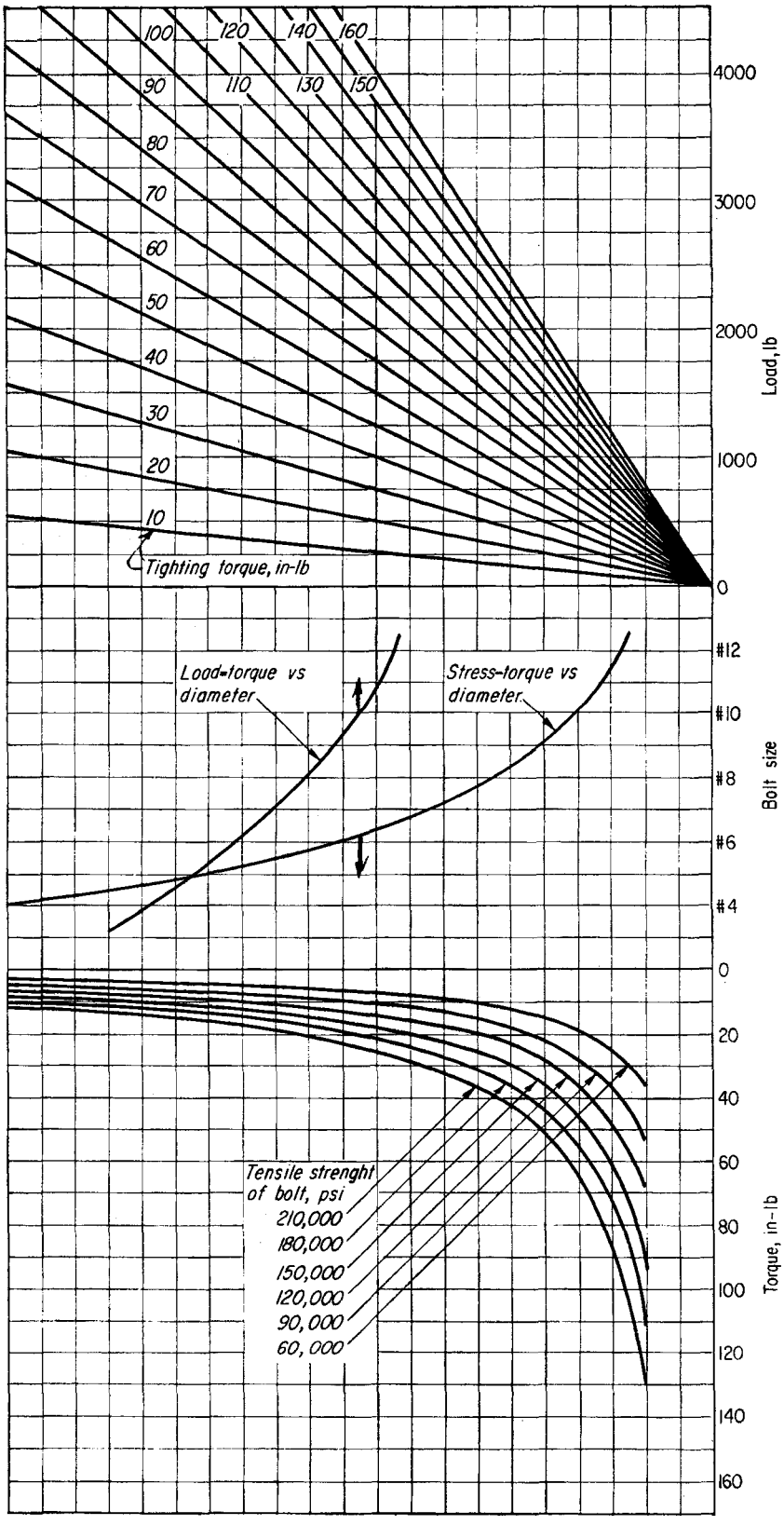
$$\frac{S_s}{0.6} = \frac{1}{2}\left[(1+0.3)\sqrt{4\,(C\,S_s)^2}\right],\quad C = 1.3$$

Eq 1 now becomes

$$S = \frac{1}{2}\left[(1-\epsilon)\,S_t + (1+\epsilon)(S_t^2 + 6.76\,S_s^2)^{1/2}\right]$$

Substituting values for S_t, S_s/S_t, and $\epsilon = 0.3$, and dividing by T gives

$$\frac{S}{T} = \frac{0.89 + 1.66\sqrt{1 + (5.2\,m\,d_p/d_r)^2}}{d_r^2\,(\nu D + m\,d_p)} \qquad (2)$$



2..FOR BOLT SIZES, #4 TO #12.

the ultimate (tensile) strength of the bolt, which results in a maximum permissible prestress of 80 × 85 = 68%, or roughly two thirds, of the tensile strength. However, a value of only 60% of the tensile strength was used in the charts to compensate for certain deviations, which will be explained later.

An ideal limit suggested by some researchers is that the bolt be tightened to its yield point and then slightly loosened. The drawback here is that the yield point of the bolt cannot be determined with any reasonable degree of accuracy.

Equations

The equations which form the basis of the analysis and related design curves are as follows:
Stress-torque ratio:

$$\frac{S}{T} = \frac{0.89+1.66\sqrt{1+(5.2md_p/d_r)^2}}{d_r^2\,(\nu D+md_p)}$$

Preload-torque ratio:

$$\frac{P}{T} = \frac{2}{\nu D + md_p}$$

where

$$m = \frac{\tan(\beta+\phi)}{\cos\alpha}$$

Design curves

The most common bolt applications utilize steel bolts and nuts, unlubricated (as received from the manufacturer). For this type of application, values of 0.14 for collar friction ν, and 0.12 for thread friction μ, are reasonable and were used to plot the curves. The curves are plotted for fine threads but may be used for coarse

BOLT CANDIDATES FOR PROBLEM I

BOLT DIAMETER, IN.	TIGHTENING TORQUE, IN.-LB
1/4	180
5/16	225
3/8	275
7/16	320
1/2	360

and extra-fine threads with only small error. When the recommended tightening torque from the curves for a specific bolt size is applied to coarse threads instead of fine, the prestress will be somewhat higher, probably on the order of 10%; when applied to extra-fine threads, it will be somewhat less, again probably on the order of 10%. For critical applications, the formula can be used to make calculations which can be checked from the curves.

The three groups of curves, Fig 2, 3, and 4, are for bolt sizes ranging from #4 to #12, from ¼-in. to ½-in. diameter and from ½-in. to 1-in. diameter, respectively. In each group the top family of curves is for the preload-torque ratio and the bottom family for the stress-torque ratio. The center curves act as pivot points.

Problem I—bolt selection

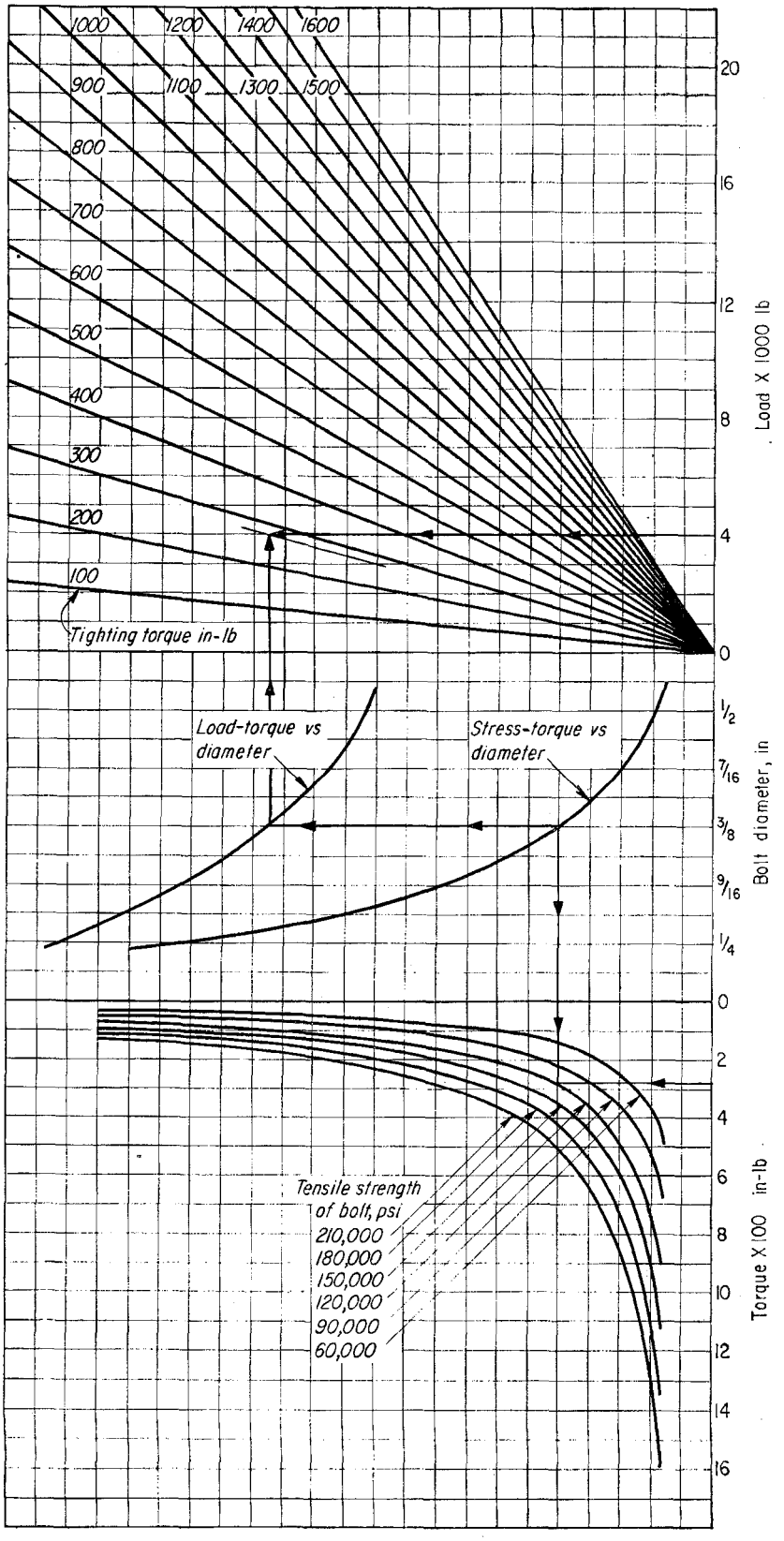
The design requirement is to select a bolt that can successfully sustain a 4000-lb axial load. Here are the steps to follow:

1) Make the preload equal to the anticipated axial load. Hence, $P = F$. (The preload can also be made larger than the anticipated axial load.)

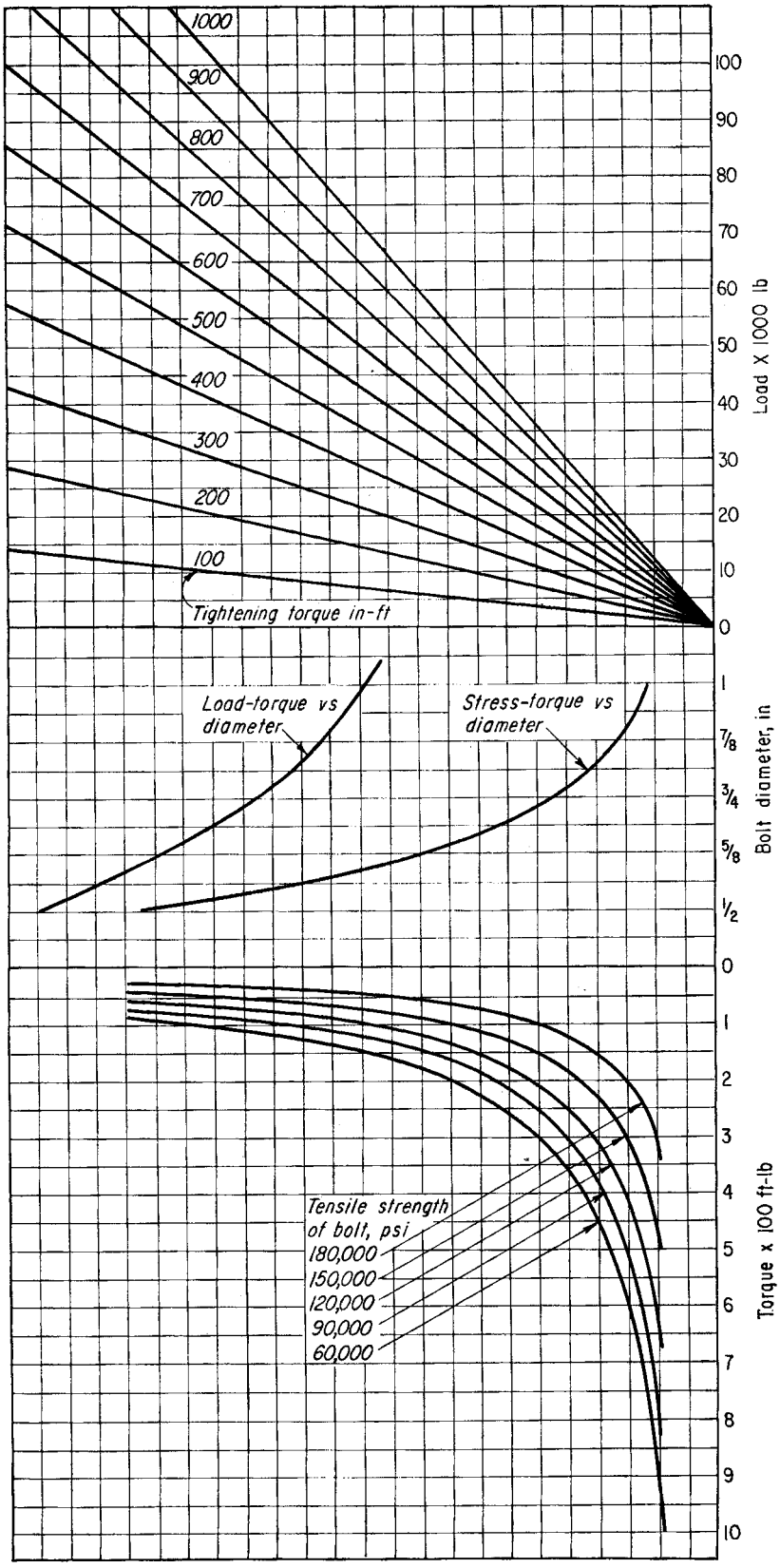
2) Select several bolt sizes in the size range expected, then check tensile-strength and tightening-torque requirements. For example, choose a bolt of ⅜-in. diameter. This calls for use of the curves, Fig 3, to the right. From the column for ⅜-in. bolt diameter, read across to the load-torque line, then pivot upward to intersect the 4000-lb load line, as shown by the arrows. A reading of the radial torque line at this point gives a tightening-torque value of 275 in.-lb.

3) Return to the ⅜-in. bolt diameter and read across to the stress-torque line, then pivot downward to intersect

TENSILE LOAD, LB	BOLT STRENGTH, PSI
4000	Over 210,000
4000	180,000
4000	120,000
4000	90,000
4000	60,000



3. FOR BOLT DIAMETERS, ¼ IN. TO ½ IN.



4..FOR BOLT DIAMETERS, ½ IN. TO 1 IN.

a line drawn from the 275 in.-lb. point. At this intersection the tensile strength is 120,000 psi. This is the material-strength requirement for the bolt.

4) The procedure is repeated for other bolt sizes and the results are listed in the table, p 64, in order of bolt size. All the bolts, with the exception of the ¼-in. bolt, will satisfy the requirements of absorbing a 4000-lb axial load without stress fluctuations or plate separation. The smaller bolts require higher-strength material but may be better for lightweight design.

Problem II—a check on accuracy

Use the equations to determine the tightening torque and required material strength for a ¾-16 UNF-2 bolt that is subjected to a 20,000-lb force.

Bolt pitch = 1/16 in., d_r (from tables) = 0.6718, d_p = 0.7029, D = 0.9375, μ = 0.12, v = 0.14, α = 30 deg.

$$\text{Sin } \beta = \frac{1}{16\pi\,(0.7029)} = 0.0283$$

$\beta = 1° 37'$

$\tan \phi = 0.12$ or $\phi = 6° 50'$

$$m = \frac{\tan (1° 37' + 6° 50')}{\cos 30°}$$

$$= \frac{0.14856}{0.86603} = 0.17154$$

$$T = \frac{P}{2}(vD + md_p)$$

$$T = \frac{20{,}000}{2}\,[(0.9375)\,(0.14) + (0.7029)\,(0.17154)$$

T = 2518 in.-lb. = 210 ft-lb

Substituting values of T, m, d_p, d_r, μ and D in Eq 2, p 63, and combining:

$$\frac{S}{2518} = \frac{0.89 + 1.66\sqrt{1 + 5.2\,(0.180)^2}}{(0.451)\,(0.131 + 0.121)}$$

$$\frac{S}{2518} = \frac{0.89 + 2.34}{(0.451)\,(0.252)} = 70{,}000 \text{ psi}$$

S = 70,000 psi

For 60% prestress, the tensile strength of the material should be:

$$S_t = \frac{70{,}000}{0.60} = 116{,}667 \text{ psi}$$

From the curves on page 66, for ¾-in. bolt and 20,000-lb load:

T = 210 ft-lb; S_t = 120,000 psi.

For **reprint** of above article, just check 709 on one of the Reader Service cards found in this issue.

World of Self-Locking Screws

Screws and bolts with self-locking ability are making the scene on more products because of today's emphasis on safety and reliability.

Frank Yeaple

Self-locking ability in fasteners is definitely in. The fear of costly lawsuits stemming from a fastener's loss of clamp load is making manufacturers of products spend a little more time and money to assure joint reliability. Hence design engineers have a freer hand in selecting locknuts and lockscrews.

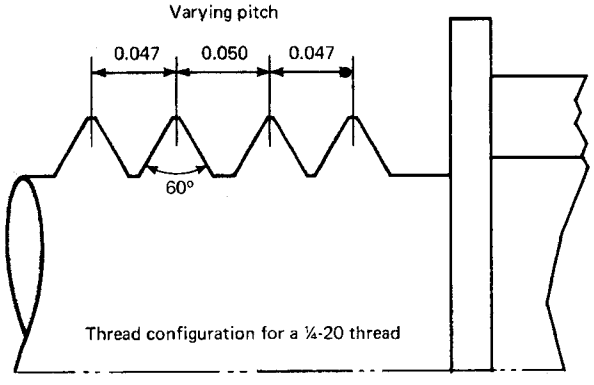
Last month's issue contained a roundup of proprietary types of locknuts. Locknuts are generally cheaper than lockscrews, and fewer special parts (the locknuts) need to be stocked to handle original and replacement parts. But because lockscrews mate with a standard nut, they can provide a better guarantee that the locking system will be retained even when the original nut is lost.

This roundup of self-locking screws and bolts includes many novel thread forms that are competing for the job of maintaining a tight joint under adverse conditions. Also included are special shank and head designs and the use of plastic inserts and anaerobics to improve self locking characteristics.

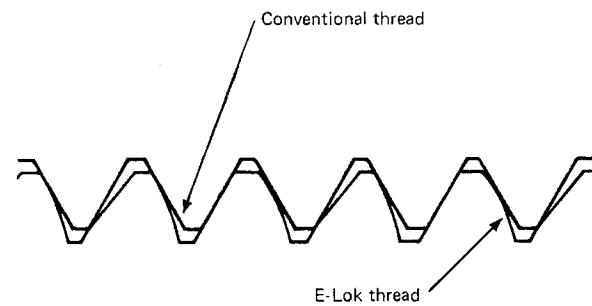
Variable pitch threads. A screw thread with an oscillating pitch distance can increase and decrease between maximum and minimum values to provide a high resistance to self loosening from shock and vibration. The Leed-Lok prevailing-torque screw (Fig 1), available from National Lock Fasteners Division of Keystone Consolidated Industries (Rockford, Ill.), uses the varying pitch to induce interference on the flanks of the threads. The locking threads need not be produced along the entire length of the screw, but only at desired points. For example, the pitch for a ½-20 thread may vary as follows: 0.047, 0.050, 0.047. This significantly increases turning friction but does not deform the mating threads beyond their elastic limit.

Tilted threads. Another way to induce interference is by deflecting and slightly deforming the threads during the thread

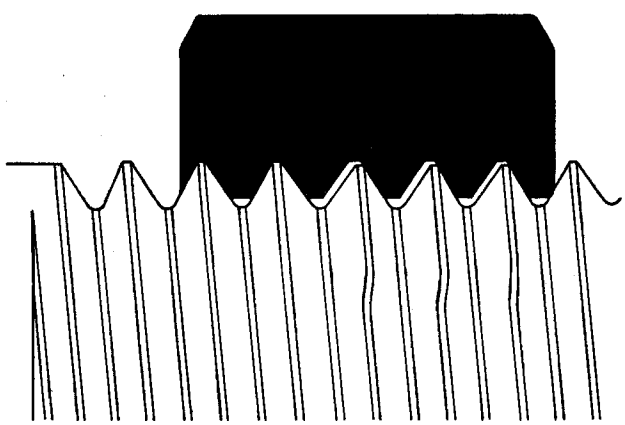
1. Leed-lok threads



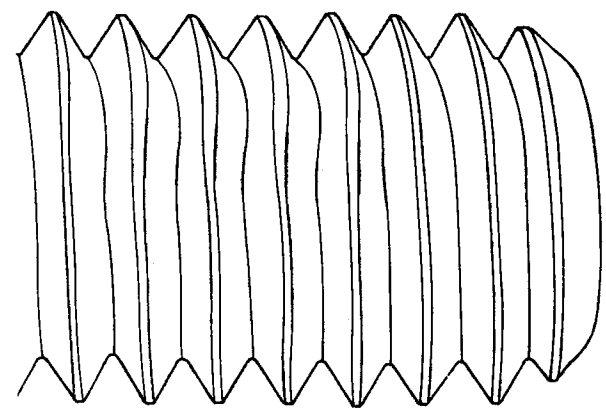
2. E-Lok threads



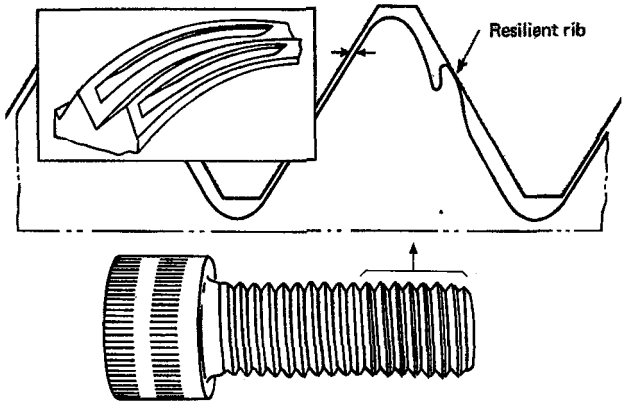
3. Vibresist threads



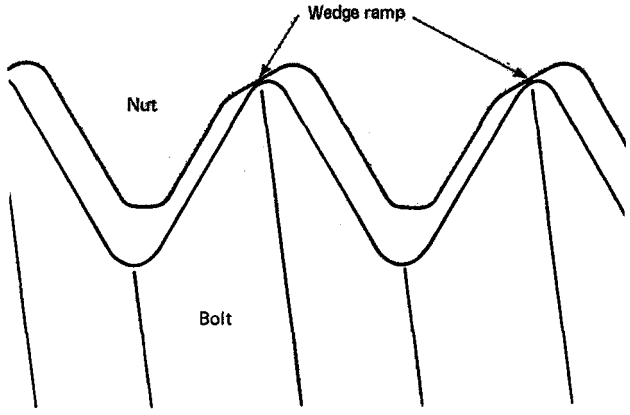
4. Tru-Flex threads



5. Orlo threads



6. Spiralock threads



rolling process when the screws are made. Employed by the Everlock Division (Troy, Mich) of Microdot on its E-Lok screws (Fig 2), the process deflects the 60-deg thread form a certain amount (about 10 deg) from the normal to the screw axis. In general, every two tilted threads are followed by two straight (normal) threads over the chosen length of screw.

Partially offset threads. In this method, portion of each thread is deformed parallel to the thread helix axis (Fig 3). As an example, Vibresist screws produced by Russell, Burdsall, & Ward (Mentor, Ohio) provide a consistent prevailing torque, slight or substantial as required. The mating internal threads wedge between the offset screw threads. Spring action is reinforced, and because locking, here as with the other altered-thread types, depends on the elasticity of metal, the screw does not lose its grip at elevated temperatures. The locking threads can be specified adjacent to the head for locking in tapped holes, along the middle of the thread length for locking in tapped holes and near the screw end for locking with standard nuts.

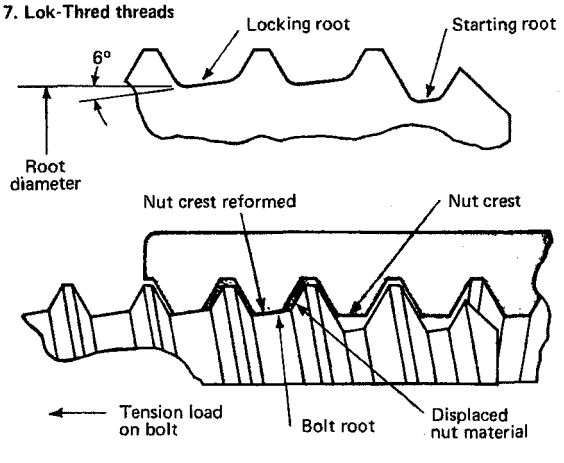
Screws with somewhat similarly deformed threads are offered by Cleveland Cap Screw (division of SPS Technologies, Cleveland, Ohio) on its Tru-Flex screws (Fig 4). In the process, threads near the end of the screw are specially deformed for a circumference of less than 180 deg of arc. When meshed with conventional female threads, the locking threads are repositioned from root to crest to induce a resisting torque. It was found that reshaping several threads within a 180-deg arc of the fastener circumference would provide ample prevailing torque with only minute changes in pitch diameter.

Resilient rib thread. The Orlo thread (Fig 5) has a cold-formed resilient rib on the non-pressure flank of the thread, either part way or continuously around. When a screw with Orlo threads is assembled into a threaded part, the ribs are compressed like springs to force the screw's pressure flanks against the mating threads, and this will increase resistance to rotational forces from vibration or shock. The spring-like action of the rib threads permits the screw to be reused effectively. Orlo threads are available on screws by Holo-Krome Co (West Hartford, Conn)., and Pioneer Screw & Nut (Elk Grove Village, Ill).

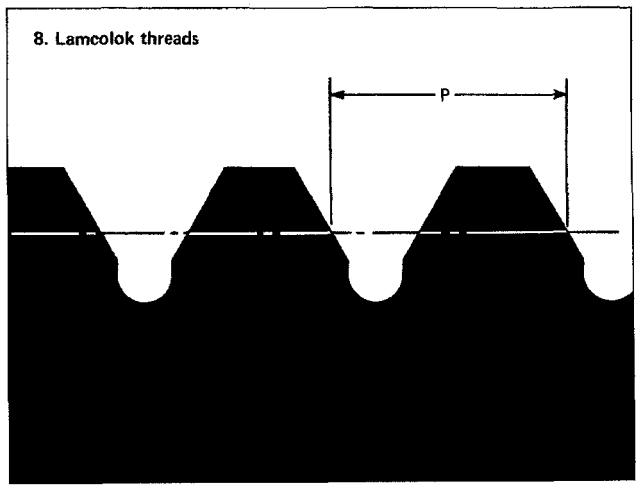
Wedge-ramp roots. This internal thread form, called Spiralock (Fig 6), is applied to locknuts and tapped holes to provide locking characteristics to standard bolts. The key innovation is the addition of a 30-deg ramp to the roots of conventional 60-deg threads. When the bolt is seated, the crest of the bolt thread is pulled up tight against the ramp and is wedged firmly with positive metal-to-metal contact that runs the entire length of the nut or tapped hole. The special thread form, in fact, allows wide latitude in bolt tolerances. Spiralock™ locknuts are available from the Greer and the Kaynar divisions of Microdot (Greenwich, Conn.); taps for producing the thread may be obtained from Detroit Tap and Tool Co. (Warren, Mich.)

Threads with special wedge-ramp roots also have been applied to screws. In the Lok-thred form (Fig 7), available from Lock Thread Corporation and National Lock Fasteners (Rockford, Ill.), the thread of the screw itself performs the locking action. This male thread is shallow, with ample radii and a wide root pitched at a

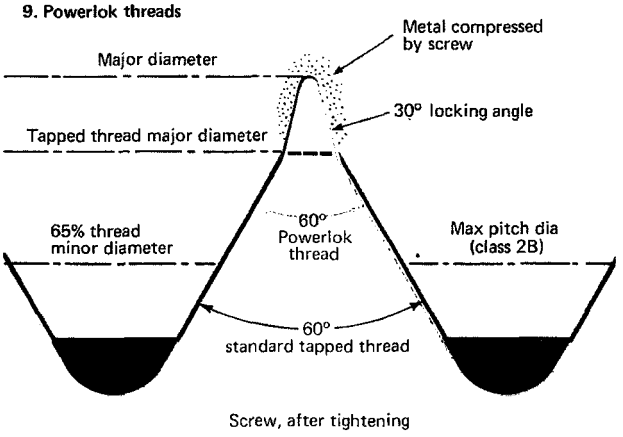
7. Lok-Thred threads



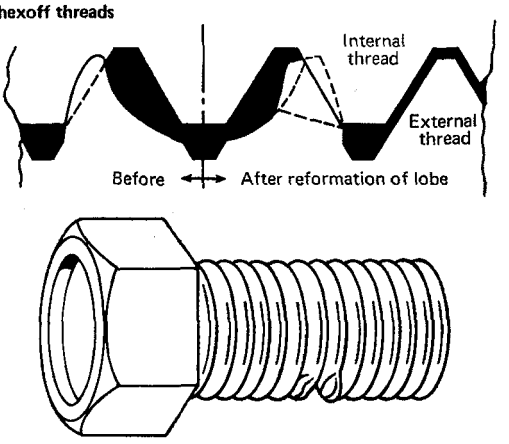
8. Lamcolok threads



9. Powerlok threads



10. Chexoff threads



locking angle. These locking roots converge toward the head. A Lok-Thred bolt enters an ordinary tapped hole freely for a few turns, then meets resistance when the bolt root contacts and, by swaging, reforms the nut threads to a perfect fit about the wedge-like locking roots of the bolt. In service, much of the clamp load is carried on the tapered roots, which wedge against the nut crest to lock securely as the work load increases. Bolts which have this type of wedge root thread have a larger root diameter than ordinary threads, which gives strength in tension, torsion and shear, as well as increased endurance limits.

Locking roots. A self-locking thread based on flank interference with standard internal thread flanks (Fig 8) provides a clamping force and holding torque that remains constant. Developed by Lamson & Sessions (Cleveland, Ohio), the Lamcolock thread design can be rolled on almost any bolt or screw. It employs a recessed full-radius root and a decreased major diameter giving the thread a wide, squat look. The decreased major diameter provides room for material to flow into the root recesses, thus averting possible galling.

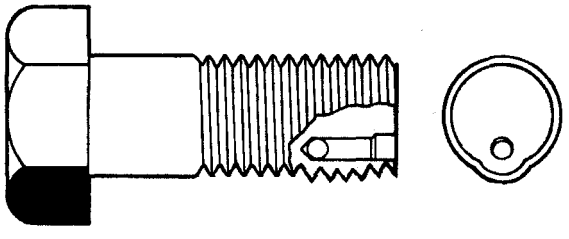
Locking crests. The locking ability of Powerlok screws (Fig 9) is enhanced through the combination of a novel 60 deg - 30 deg thread form and a tri-lobular thread body section. Locking action is developed at the outermost radius of the torque arm of the screw body, whereas most locking screws develop their resistance at lesser radius points. The deeper thread form of the Powerlok geometry, along with a slight increase in the major diameter of the thread as compared to equivalent size conventional screws, also adds to the locking ability. Basically, the nut-thread metal is elastically deformed in the compressed areas created by the 30-deg portion of the thread. Powerlok screws are available from several fastener manufacturers, including Continental Screw Co (New Bedford, Mass.), Midland Screw Co. (Chicago, Ill.), Central Screw Division of Microdot, and Elco Industries (Rockford, Ill.).

Resilient bulges. By deforming several threads on one side of the Chexoff screw to form lobes (Fig 10), controlled thread interference is induced when the screw is threaded. Available from Central Screw Co, (Des Plaines, Ill.), the special screws with lobes create a wedge-like effect by exerting pressures on the opposite side of the mating threads. The lobes may all be located on the same line or else staggered in order to provide considerable periphery pressures.

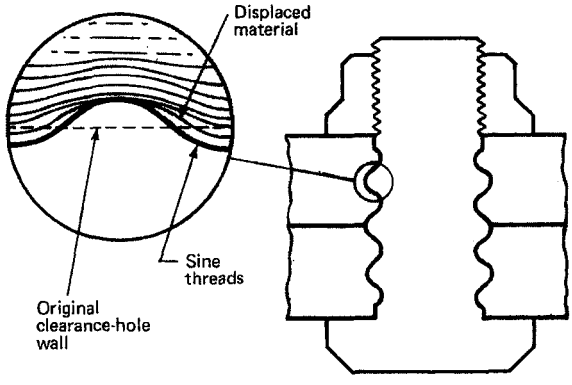
In another design, Deutsch Fasteners Corp. induces a resilient bulge on one side of the bolt (Fig 11) that increases in a similar manner the frictional contact between mating threads on the opposite side of the bolt. The bulge is formed by the interference action of a precision ball, pressed into a hole drilled close to the minor diameter of the threads during manufacture.

Sine-wave threads. In another approach to improving the locking characteristics of bolts, Valley-Todeco, Inc. (Sylmar, Calif), has developed its Sine-Lok interference-type thread (Fig 12) for use on the upper regions of a bolt, normally consisting merely of a straight shank. The lower portion of the bolt has conventional threads. During assembly, the bolt shank,

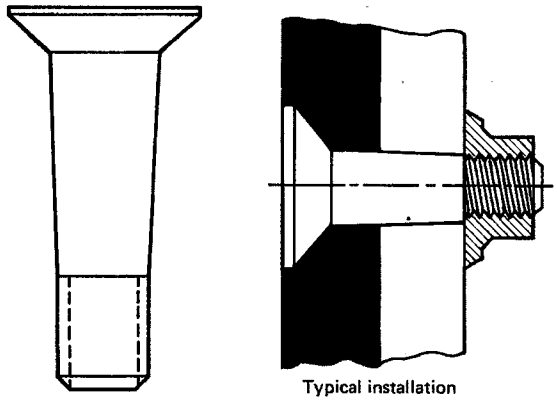
11. Resilient-bulge threads



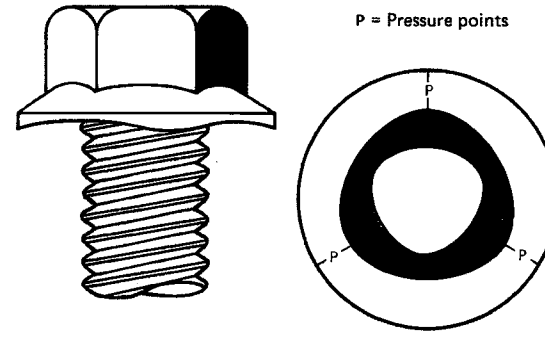
12. Sine-Lok bolt shank



13. Taper-Lok shank



14. Uniflex head



which has a series of modified sine waves, is threaded through the clearance holes of the two parts being assembled, instead of simply being pushed through. The sine wave threads displace the material of the clearance-hole wall into their roots. The protruding, threaded end of the bolt is then tightened by a standard nut. The bolt thus gets a double threading action that helps prevent loosening.

Tapered shank. Another way to increase the shank's grip of the parts being assembled together is to provide a slight taper to the shank, as shown in the Taper-Lok screw by Voi-Shan Division of VSI Corp. (Pasadena, Calif.). (Fig 13). Although only 0.25 in. per linear foot, the taper provides a controlled interference fit that compresses the material of the joint elastically around the hole to induce an excellent preload condition.

Head-grip screws. Many new head shapes for screws are designed to increase resistance to vibrational loosening in joints. The Uniflex head (Fig 14), developed by Continental Screw Co. (New Bedford, Mass) to complement the company's line of trilobular thread-rolling screws, has a washer-like, undulating head-bearing surface. When the surface, which has three alternate high-and-low areas, is tightened against the joint being assembled, the relieved areas are aligned with potential stress points at the lobes of the trilobular screw thread. As a result of this bearing-area relief, thread engagement is increased and bolt loosening noticeably resisted.

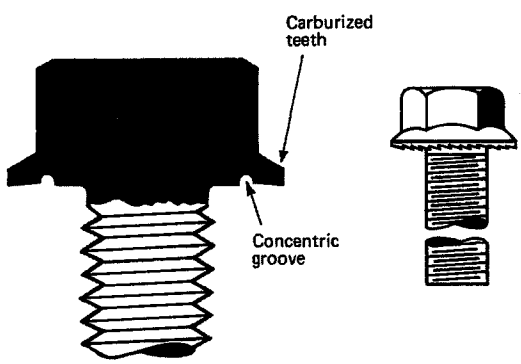
High locking power and clamping force are provided with Tensilock screws available from Eaton Corporation (Massilon, Ohio) (Fig 15). The screw head has a concentric circle of 24 embedded, carburized teeth, with an outer concentric groove that permits flexing of the head to occur. The Durlok fastener available from SPS Technology's Cleveland Cap Screw (Cleveland, Ohio) also has ratchet-like teeth around the periphery of the bearing surface (Fig 16). To limit depth of penetration and marring of the mating surface, the serrations are encircled by a smooth outer bearing area.

Lockscrews with the toothed portion of the head furnished in the form of a preassembled washer include the Melgrip screw by Elco Industries (Rockford, Ill) (Fig 17), and Sems screws available from a number of manufacturers, including Shakeproof Division of Illinois Tool (Elgin, Ill), National Lock Fasteners (Rockford, Ill.), and Central Screw (Fig 18). Melgrip's locking effectiveness results from the mating serrations on the underside of the bolt head and top of the washer surface, coupled by the bidirectional gripping teeth on the periphery of the washer which embed into the joint material. Thus, the washer cannot skid or score.

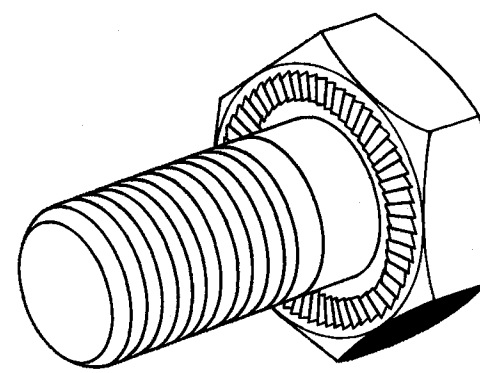
Sems screws are available in a vast variety of washer types to provide spring tensioning for improved loosening resistance, as well as to bridge oversized holes or insulate and protect material surfaces.

Nylon-pellet insert. Self-locking screws that use a resilient nylon insert in the threaded section to develop a

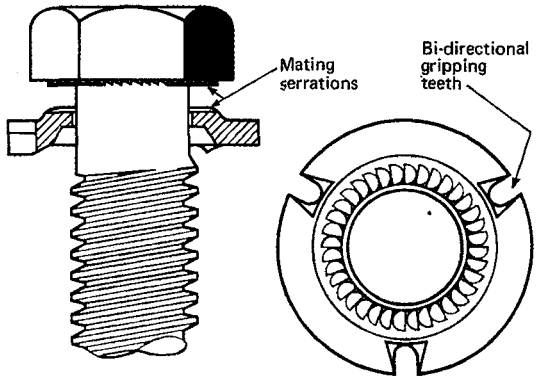
15. Tensilock head



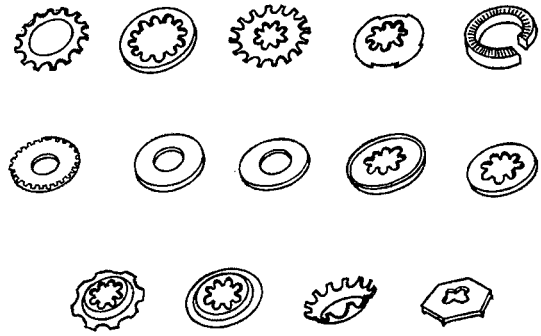
16. Durlok head



17. Melgrip washer-head



18. Sems washer types



prevailing-torque locking action (Fig 19) have been a familiar product for a number of years. The nylon pellet, press-fitted into the screw, projects slightly beyond the crest of the thread. Once the threads are engaged, the screw is held in position by lateral pressure. The pellet technique can be applied also to nuts and studs. Screws with nylon-pellet inserts are available from Nylok Fastener Division of USM Corp. (Paramus, N.J.) and ND Industries, Inc. (Troy, Mich.).

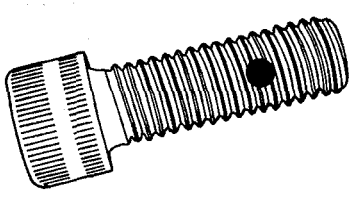
Nylon fused patch. Another effective and well-established way to help a screw resist loosening is by use of a plastic locking patch that is fused on a dimensionally controlled area of the screw threads, shown in Fig 20. This nylon patch is thickest in the center and feathers-out along the edge to provide a gradual engagement of the locking patch as it encounters mating threads. As the mating threads fully engage the patch, the nylon is compressed to build up a resistance to turning at the right in the figure, and a strong metal-to-metal contact between threads at the left. This type of screw is available from the Esna Division of Amerace Corp. (Union, N.J.), Long-Lok Fasteners Corp. (Cincinnati, Ohio), Holo-Krome Co., (West Hartford, Conn.), and the Unbrako Division of SPS Industries (Jenkintown, Pa.)

Adhesive thread locking. Epoxy and anaerobic adhesives and stiffly viscous fluids have become popular for turning low-cost plain bolts and nuts into lock fasteners. Epoxy, which is a two-part adhesive, is applied in the form of alternating strips or microcapsules to the threaded fastener. Once applied, the epoxy (or one of the anaerobics) remains dormant on the fastener (Fig 21) until a cure is activated by engagement with a mating thread. The curing process for most of the chemical adhesives may continue for days, although by the twelfth hour a good bond has generally been achieved. The adhesive technique, however, does not offer reusability capabilities. Screws with preapplied adhesives are available from ND Industries, (Troy, Mich.), Cleveland Cap Screw (Cleveland, Ohio) and Camcar Division of Textron (Rockford, Ill). Or you can buy your own.

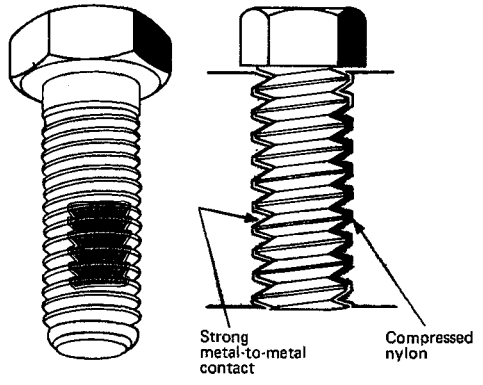
Highly viscous fluid coatings, such as Vibra-Tite available from ND Industries (Troy, Mich.) and Oakland Corp., offer a compromise by making the parts adjustable as well as improving self-locking. The user can apply it from a bottle with a brush applicator, like glue. Vibra-Tite, however, is not an adhesive, so its primary usage is not to provide a heavy-duty locking capability; but it does permit the fastener to be assembled, disassembled or adjusted.

Prevailing torque. Remember that most self-locking screws rely on friction in one form or another to hold fast against axial or transverse vibration. Transverse vibration is most difficult to protect against because accelerating forces can cause momentary slip at a microscopic level, eventually loosening the thread. Lab and field tests are recommended.

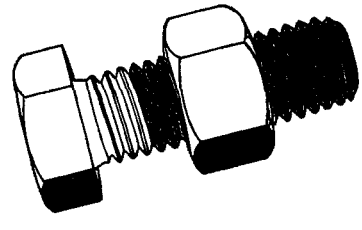
19. Nylon-pellet screw



20. Nylon locking-patch screw



21. Adhesive-lock screw



Unconventional Thread Form Holds Nut to Bolt During Severe Vibrations

Wedge shape at root of nut creates metal-to-metal contact over mating thread length. The result is a firmer grip.

When subjected to high transverse vibrations, fasteners tend to loosen with all the undesirable consequences that follow. Engineers at Microdot Inc. (Troy, Mich.) say they have solved the problem. Their solution: a new thread that resists vibrations more effectively than others now available.

Known as Spiralock, the thread is being used in a free-spinning, flanged nut, to eliminate internal self-loosening, the cause of walk-off. The thread form incorporates a wedge ramp at the root of the thread. This wedge ramp is created by widening a portion of the normal thread angle at its root.

Hold tight. When torque is applied, the crests of the mating fastener are pulled against the wedge ramp to form metal-to-metal contact over the entire length of the spiral path of mating threads. Such contact prevents the fastener from loosening under severe vibration conditions.

According to Microdot designers, the fastener works effectively because under clamp load the air space between the mating threads is eliminated. Air space, which is normal in free-spinning fasteners, is the cause of internal self-loosening, they contend.

With such fasteners, the diametrical tolerances needed on both the male and female threads frequently create air spaces of 0.018 in. The average air space is 0.010 in. and the minimum air space needed for assembly is 0.0025 in.

Thus, when any assembly using a regular free-spinning nut is subjected to transverse vibration, the air space allows the mated thread flanks to move laterally and the nut walks-off. Laboratory tests at Microdot on a Junkers vibration testing machine compared performance of Spiralock with other nuts.

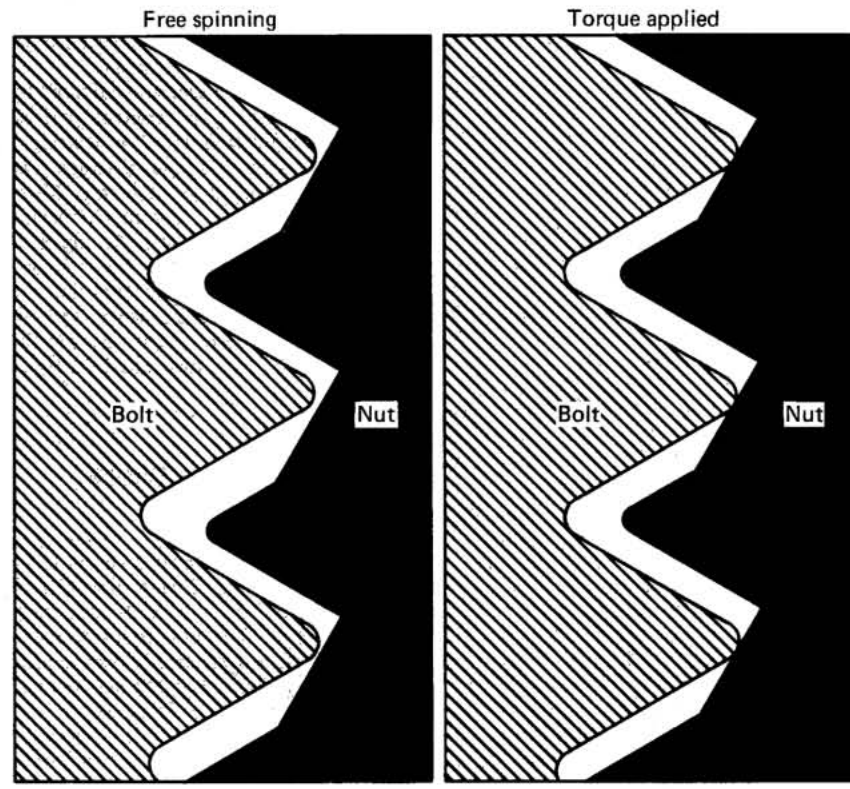
Nuts of various sizes were tightened to 75% of proof load on standard nuts and then subjected to high amplitude vibration. In every case, the test engineers say, the bolt failed from metal fatigue before the nut loosened. All bolts and nuts used in the test were SAE grade 8.

When comparing Spiralock nuts to prevailing torque nuts several differences should be kept in mind, the designers say. Under normal vibration conditions, both Spiralock and prevailing torque nuts retain an acceptable amount of initial clamp load when tightened to 75% proof load. Under high vibrations the new wedged thread retains a higher percentage of initial clamp load than a prevailing torque nut.

However, initial clamp load must be achieved for Spiralock to be effective. In applications where there can be no assurance of achieving initial clamp load or if it can be lost due to brinelling, the new nut should not be used, the designers warn.

Where it can be used, though, the Spiralock offers important benefits to manufacturers. It is free-spinning, reusable, and speeds assembly time. Above all, its ability to withstand high vibration or cyclic transverse loading qualify it for high-vibration applications, it is said.

Typical free-spinning flanged lock nut



Widened angle at thread root forms wedge. Bolt remains free-spinning until torque is applied. Then bolt threads contact wedge, eliminate air space and prevent lateral movement.